Variable-speed centrifugal compressors are used in a wide variety of industries and applications, including natural gas pipelines and processing plants, oil refineries and chemical plants, air-separation units, refrigeration and air conditioning equipment refrigerant cycles, gas turbines and auxiliary power generators, and many more. Some natural gas plants, for example, employ a propane-precooled, mixed-refrigerant natural gas liquefaction process consisting of a classical propane liquefaction cycle that precools both the natural gas feed and the mixed refrigerant followed by a mixed-refrigerant liquefaction cycle that provides the low-temperature refrigeration needed to liquefy the natural gas. Compression is a key element of both refrigeration cycles.

Dynamic process simulation offers many tangible and intangible benefits (1). Integrating the turbomachinery controls (TMC) into the dynamic simulation of a compressor system enables the engineer to select properly sized equipment that can cope with transient conditions (surge conditions in particular), reduces commissioning time through pre-tuning of the regulatory controls, improves operator training by exposing operators to realistic simulated operating scenarios, and provides verified startup and shutdown procedures during site acceptance testing before startup takes place (2).

Dynamic simulators for modeling processes that involve gas compression require a rigorous unit-operation model of a centrifugal compressor. The performance maps provided by compressor manufacturers are a critical element of this model. This article explains how to use the performance maps for variable-speed, fixed-inlet guide-vane centrifugal compressors for the dynamic simulation of a compressor system.

Performance maps and their use in simulation

A performance map describes how a compressor’s polytropic head and power vary with volumetric suction flowrate and rotational speed for a specific set of suction conditions (i.e., fluid molecular weight, pressure, temperature, compressibility, and isentropic exponent at the inlet of the compressor). Manufacturers typically provide performance maps as two sets of performance curves — a series of plots of polytropic head vs. volumetric suction flowrate, and a corresponding series of power vs. flow plots. Each pair of curves corresponds to a different rotational speed, and all pertain to the same suction conditions.

Performance maps may be constructed for more than one set of suction conditions. Figure 1 presents two performance maps for a high-pressure mixed-refrigerant compressor for a natural gas liquefaction process. The top performance map corresponds to design operation, and the bottom map corresponds to derime operation, which removes rime (the granular ice that forms when supercooled droplets freeze rapidly on contact with a cold surface) from the compressor. These maps were generated from data obtained by manually digitizing the performance maps provided by the manufacturer; the irregularities in the curves are a result of the manual digitization process.

Process simulators combine the performance maps with material and energy balances and thermodynamic relationships to predict the performance of the compressor under various operating scenarios. However, the performance maps supplied by the manufacturer are not broadly useful for process simulation because, strictly speaking, they apply only at the suction conditions and rotational speeds for which they
were constructed. Process simulation requires prediction of the compressor’s performance not just at these suction conditions and speeds, but over the compressor’s entire operating range, from surge to stonewall, at varying suction conditions and rotational speeds.

Some process simulators ignore the fact that the performance maps relate to specific suction conditions, or allow users to enter multiple performance maps in tabular form and synthetically transition between them as the suction conditions change. However, as shown in Figure 2, the performance maps for different suction conditions can be quite different. In this instance, the scales for power differ by more than an order of magnitude. Using the performance map for design operation at all operating conditions will certainly yield poor results for derime operation.

Varying rotational speed can be handled somewhat better than varying suction conditions, for example, by linear interpolation and extrapolation on an irregularly spaced nonrectangular grid. Although linear interpolation can yield reasonable results, linear extrapolation breaks down as the rotational speed moves further away from the lowest and highest speeds for which performance curves were constructed. This limits the utility of the performance maps to normal operating speeds.

\[\text{Figure 1. These performance maps represent design (top) and derime (bottom) operation of a high-pressure, mixed-refrigerant compressor.}\]

\[\text{Figure 2. Superimposing performance maps reveals that the power vs. suction flowrate relationships for design and derime operation are very different.}\]
Computational Methods

If the unit-operation model of the compressor does not adequately account for variation in the suction conditions and rotational speed, the predictions of the process model in which it is used — for example, the speed and shaft power needed to achieve a specified (controlled) discharge pressure — will certainly be flawed, and may even be misleading. If the process model is used in a dynamic simulation study, the trajectories for the operating variables will be incorrect, possibly significantly, which could lead to false conclusions. Simulation-based training using a model that misrepresents the actual process behavior could cause an operator to make incorrect operating decisions.

Transforming the performance maps

The challenge, therefore, is to find a way to make the performance maps applicable over the entire range of suction conditions and speeds encountered in the operation of the compressor. This can be achieved by transforming the performance maps to a coordinate system that is independent of both suction conditions and rotational speed. This transformation involves two steps.

First, dimensional analysis is used to construct a coordinate system that is independent of suction conditions (3). A key aspect of this analysis is the identification of a set of dimensionless parameters in which the pertinent forces can be accurately represented.

Functional relationships are set up in which polytropic head \( H_p \) and shaft power \( P \) are the dependent variables and volumetric flowrate \( Q \) and rotational speed \( \omega \) are the independent variables. The suction conditions and the compressor size are incorporated into the functions by including viscosity \( \mu \), density \( \rho \), and acoustic velocity \( a \) at the suction of the compressor and the diameter \( d \) of the compressor as additional independent variables. These functional relationships are represented as:

\[
H_p = f_3(Q, \omega, \mu, \rho, a, d) \tag{1}
\]
\[
P = f_1(Q, \omega, \mu, \rho, a, d) \tag{2}
\]

Each of these relationships involves seven dimensional parameters. Using the Buckingham Pi Theorem and assuming turbulent flow, they can be simplified to:

\[
H_{pa} = f_3(Q_n, \omega_n) \tag{3}
\]
\[
P_n = f_1(Q_n, \omega_n) \tag{4}
\]

where the subscript \( n \) denotes a dimensionless variable.

Although this dimensionless coordinate system was developed for compressor control, it is equally applicable to process simulation.

Next, a modified form of the Fan Laws is applied to further transform the dimensionless coordinate system to one that is also independent of rotational speed. The Fan Laws of centrifugal pumps or fans, also known as the Affinity Laws, express the influence of changes in rotational speed, diameter, and density on volumetric flowrate, head \( H \) (or pressure), and shaft power. When the fan diameter and the fluid density at the fan suction are constant, the Fan Laws can be expressed as:

\[
\frac{Q_1}{Q_2} = \frac{\omega_1}{\omega_2} \tag{5a}
\]
\[
\frac{H_1}{H_2} = \left( \frac{\omega_1}{\omega_2} \right)^2 \tag{5b}
\]
\[
\frac{P_1}{P_2} = \left( \frac{\omega_1}{\omega_2} \right)^3 \tag{5c}
\]

It follows from these relationships that when \( H/\omega^2 \) and \( P/\omega^3 \) are plotted against \( Q/\omega \), the performance curves for a given set of suction conditions reduce to a single curve for head and a single curve for power. However, because the Fan Laws do not strictly apply to compressors, a modified form of the Fan Laws must be used.

The resulting coordinate system, referred to as the reduced coordinate system, has as its dependent variables the transformed (reduced) polytropic head \( H_{pr} \) and reduced power \( P_r \), and as its independent variable the reduced volumetric suction flow \( Q_r \). In the reduced coordinate system, the performance curves for polytropic head and for power at different suction conditions are each represented by a single
aggregate curve, as shown in Figure 3. These aggregate curves are expressed as:

\[ H_{r} = f_4(Q_r) \]  \hspace{1cm} (6)
\[ P_r = f_5(Q_r) \]  \hspace{1cm} (7)

Transforming the performance maps to the reduced coordinate system is very powerful. Consider the following modeling scenario. Given the suction conditions, rotational speed, and volumetric suction flow for a particular operating condition (real or simulated), the reduced volumetric suction flow is calculated by applying the transformations, calculating first \( Q_n \) and \( \omega_n \), and then \( Q_r \). The reduced polytropic head and reduced power corresponding to the reduced volumetric suction flow are then determined from the aggregate performance curves, \( i.e., \) from \( f_4(Q_r) \) and \( f_5(Q_r) \), respectively. The polytropic head and power in engineering units are then easily calculated by inverting the transformations, first calculating \( H_{p,n} \) and \( P_{n} \), and then \( H_p \) and \( P \). Transforming the performance maps provided by the compressor manufacturer, which strictly apply only at their corresponding suction conditions and rotational speeds, to reduced coordinates in this way enables performance curves for polytropic head and power that are particular to different operating conditions \( i.e., \) suction conditions and rotational speed \) to be constructed.

To make a reduced performance map \( i.e., \) the aggregate curves of reduced polytropic head and reduced power \) more useful, the aggregate curves need to be represented by appropriate functions whose parameters are fitted to the reduced data.

**Functional representation of the reduced performance map**

Based on a visual inspection of Figure 3, a quadratic function appears to be a good choice for representing the aggregate performance curves for both polytropic head, \( f_4(Q_r) \), and power, \( f_5(Q_r) \). However, experience has proven otherwise. A quadratic function is not a suitable representation of the aggregate power curve, which is not symmetric about the perpendicular line through its apex. In addition, a quadratic function fitted to the data typically will not extrapolate through the origin — a necessary physical constraint. A quadratic function is also not a suitable representation of the aggregate curve of polytropic head at low values of reduced volumetric suction flow, where it becomes nearly linear.

The next likely choice of functional form would be a higher-order polynomial. However, high-order polynomials can wobble, even within the range of the data being fitted, and they may not extrapolate well — both significant disadvantages for process simulation.

Experience has shown that a very good fit is obtained when the aggregate performance curves are divided into three segments — representing the surge, normal, and stonewall regions of the performance curves — and the segments are represented by the functions listed in Table 1.

The functions’ parameters and the segments’ end points are determined by fitting the curves to the reduced data. This yields smooth, piecewise-fitted curves, as shown in Figure 4. The functions fit the curve segments well, and they extrapolate well for both low and high reduced volumetric suction flows. Although flowrates below the reduced surge flow are not physically meaningful, reasonable extrapolation to low flows is required to ensure robust solution of the compressor unit-operation model. Reasonable extrapolation in the stonewall region facilitates robust solution up to the stonewall flow.

**Table 1. The aggregate performance curves can be divided into segments, each represented by a different functional form.**

<table>
<thead>
<tr>
<th>Curve Segment</th>
<th>Polytropic Head</th>
<th>Power</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surge</td>
<td>Exponential</td>
<td>Quadratic</td>
</tr>
<tr>
<td>Normal</td>
<td>Power</td>
<td>Power</td>
</tr>
<tr>
<td>Stonewall</td>
<td>Quadratic</td>
<td>Quadratic</td>
</tr>
</tbody>
</table>

**Figure 4.** The surge, normal, and stonewall operation segments of the aggregate performance curves can be represented by different functions.
Computational Methods

Using the reduced performance map in dynamic process simulation

As stated previously, accurately simulating the behavior of processes that include centrifugal compressors requires process simulators that rigorously model these devices over their entire range of suction conditions and rotational speeds, from surge through stonewall operation.

Figure 5 is a typical compressor loop flowsheet developed using such a simulator. Figure 6 presents details for compressor unit operation K-100, which has been configured as a centrifugal compressor using the reduced coordinate system representation of the performance maps. The polytropic head performance map for this compressor, Figure 7, was constructed from the reduced aggregate polytropic head curve at the existing suction conditions, at various speeds. The current operating point, represented by the red dot, is superimposed on the performance map. In dynamic simulation, the operating point moves as the suction conditions, rotational speed, and/or discharge pressure vary. For the purpose of illustration, a transfer function (the K-100 speed block of the flowsheet) was applied to cause the rotational speed to oscillate, as depicted by the green line on the strip chart in Figure 8. The red and blue lines in Figure 8 represent the dynamic responses of the volumetric suction flow and the polytropic head.
In addition to integrating reduced performance maps into its mathematical model of the centrifugal compressor unit operation, it is crucial that the process simulator also provide a tool to assist the engineer in transforming the digitized performance curves into the reduced coordinate system, that is, in creating the aggregate performance curves and fitting the parameters of the functions that represent these curves. Without such a tool, this task would simply be too daunting, and the value associated with the reduced coordinate system representation of the performance maps would not be realized.

**Figure 8.** This strip chart records the dynamic responses of polytropic head (blue) and volumetric suction flowrate (red) to variations in rotational speed (green).

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**Literature Cited**


**Grant Stephenson** is an Engineering Fellow with Honeywell’s Automation Control Solutions business, where he serves as the global process simulation architect for Honeywell Process Solutions. He has worked in the field of process simulation for more than 35 years, with particular interest in dynamic simulation, equation-oriented modeling and simultaneous solution of flowsheet models, and the application of modeling and optimization to plant operations. He is the originator of the dynamic simulation engine of the Shadow Plant simulator and is a pioneer of the hybrid solution architecture and its application to large-scale dynamic simulation. Before joining Honeywell, he held positions with DuPont Canada, Atomic Energy of Canada, the Univ. of Western Ontario (in the engineering faculty's Systems Analysis Control and Design Activity, SACDA, group), and SACDA Inc. He holds an MSc degree in applied mathematics from the Univ. of Western Ontario.